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TEST OF AUTOMATIC TEMPERATURE CONTROLLED HYDROSTATIC FAN DRIVE

U. S. ARMY TANK-AUTOMOTIVE COMMAND LABORATORIES

August 1975

by R.P. Moncy

GENERAL ELECTRIC

ORDNANCE SYSTEMS
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TACOM

PROPULSION SYSTEMS LABORATORY

U.S. ARMY TANK AUTOMOTIVE COMMAND Warren, Michigan

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TEST OF AUTOMATIC TEMPERATURE CONTROLLED HYDROSTATIC FAN DRIVE

Final Engineering Report

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In order to improve vehicle fuel economy and performance, it is desirable to employ a cooling fan system that will operate independently of engine speed and react solely to heat rejection requirements. This report describes the design, testing, and development of a temperature-controlled hydrostatic fan drive system intended for cooling engine and transmission oil in a military tracked vehicle. The testing program demonstrated satisfactory system performance and durability characteristics.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

1.0 ABSTRACT

In order to improve vehicle fuel economy and performance, it is desirable to employ a cooling fan system that will operate independently of engine speed and react solely to heat rejection requirements.

This report describes the design, testing, and development of a temperature controlled hydrostatic fan drive system intended for cooling engine and transmission oil in a military tracked vehicle.

The testing program demonstrated satisfactory system performance and durability characteristics.

2.0 INTRODUCTION

Cooling requirements of new military vehicles have created a need for more efficient and compact fan drive systems. The temperature controlled hydrostatic fan drive system satisfies that need. This report describes the design and development of a temperature controlled hydrostatic fan drive and the testing to demonstrate its performance capabilities.

The fan drive was not developed for a specific vehicle application, but was designed to provide acceptable performance for, and to be evaluated in, any of several vehicle test rigs. At the start of the program, a study was made of fan drive requirements of 11 different Army vehicles to define a hydrostatic fan drive system with maximum interchangeability between vehicles. The results of that study indicated that a single set of fan drive components would not be practical for all installations; however, maximum interchangeability could be attained using two different motor sizes with a common pump and control. The results were presented to TACOM and the decision was made to build hardware to test the fan drive system with two motor sizes, 1.25 in^3/rev motors to drive 26 HP at 6000 rpm fans and 1.00 in^3/rev motors to drive 20 HP at 8000 rpm fans. An 8.7 in^3/rev variable displacement pump was selected to drive the motors. The pump was sized to provide adequate flow for two motors in parallel plus hydraulic power for auxiliary equipment.

The fan drive system was tested in the GEOS transmission laboratory. Extensive development and performance tests were conducted with the 6000 rpm system driving two Joy Vaneaxial fans. System performance of the 8000 rpm drive was evaluated using Task Axivane fans.

3.0 OBJECTIVES

The object of this contract was to demonstrate the concept of controlling a dual fan hydrostatic fan drive system in proportion to the temperature of two oil coolant lines and independently of engine speed. Both the 6000 rpm and 8000 rpm systems would turn from minimum to maximum speed between the temperatures of 170°F and 210°F. A 100-hour system evaluation was conducted with the 6000 rpm system and system performance of both fan systems was assessed based on the following points:

1. Demonstrate temperature/speed characteristics and demand cooling,
2. Determine if a flow divider is required to balance fan speed in a dual fan system,
3. Determine if fan speed overrides are necessary,
4. Determine if system damage can occur due to fan stall,
5. Determine speed and pressure characteristics during engine speed transients at high overdrive ratios,
6. Determine system efficiency, and, in addition,
7. Investigate feasibility of utilizing a larger fan drive pump to provide auxiliary power for accessories (e.g., turret, suspension, air compressor, bulldozer, etc.).

4.0 CONCLUSIONS AND RECOMMENDATIONS

4.1 Conclusions

1. The design, fabrication, and test of this temperature controlled hydrostatic fan drive demonstrated the soundness of the concept and the performance capabilities of a demand cooling system.
2. Flow dividers are not required to maintain speed balance when two or more fans are driven in parallel from one pump. The torque-speed characteristics of the fan and the resulting pressure-flow characteristics of the fan motors act to keep the fan speeds in balance.
3. Fan speed overrides are not necessary with this fan drive control. Fan speed is regulated by motor flow independently of fan torque and, if fan speeds are unbalanced by some unforeseen cause, the speed of the fastest fan is automatically controlled. Back-up protection against overspeed is provided by the servo relief valve.
4. The servo relief valve prevents damage to the fan drive if both fans are stalled. If only one fan is stalled, the operable fan is controlled at the normal speed-temperature schedule. With normal cooling of the fan drive oil, the system can be operated continuously with one or both fans stalled, without damage to the system.
5. Acceleration tests with the drive at maximum overdrive ratio demonstrated that the control response is fast enough to prevent excessive pressure or fan speed overshoot.
6. The temperature controlled fan drive system will require less PTO power than the direct drive fan for all operating conditions other than the high temperature, high tractive effort operation. With less

fan drive power required, vehicle acceleration and high speed operation will be improved.

7. The fan drive pump has reserve capacity that could be used for auxiliary power, but, unless the pressure characteristics of the auxiliary are compatible with the fan motor pressure, the added control complexity and reduction in system efficiency could limit its use.

4.2 Recommendations

1. A system test should be performed in an actual vehicle application to evaluate the performance of the fan drive system in a closed loop cooling system.
2. A fan drive system design should be directed toward a specific vehicle application rather than a universal application. Although some applications would probably prove to be satisfactory, the diverse cooling requirements of the vehicle group would inevitably lead to poor system efficiency.
3. A unique vernatherm should be developed to satisfy the exact temperature control requirements of the fan drive system.

5.0 SYSTEM AND COMPONENT DESCRIPTION

5.1 System Description

A system description with brief functional information is included in this section; a detailed description of the pump, controller, motors, and thermostats follows in Section 5.2.

The temperature controlled hydrostatic fan drive system is shown schematically in Figure 1 and consists of the following items:

- variable displacement pump,
- hydromechanical controller assembly,
- two fixed displacement hydraulic motors,
- two thermostat assemblies,
- two push-pull control cables,
- motor line check valves, and
- oil make-up and fan drive cooling circuit (including heat exchanger and filter).

Figure 2 shows the actual test cell installation of the above components. The system layout allows a great deal of packaging latitude by close coupling the motor/controller and remotely mounting the thermostat assemblies and fan motors.

The heart of the system is the variable displacement pump which supplies a particular output flow to the fan motors according to stroking signals from the controller assembly. It is this regulated flow which determines the speed of the fan motors.

The controller communicates its pump stroking signal via a pilot valve. This signal is the product of the mechanical action of the temperature and speed sensor inputs upon the summing bar.

The position of the temperature input is a function of the temperature at the remote thermostat. When the temperature at either thermostat reaches 170°F, a mechanical signal is communicated to the controller via a precision push-pull cable. This signal is proportional to temperature until it reaches its full stroke at 210°F. In order to exercise exact control over the controller assembly during the initial development testing, the thermostats were not incorporated. Instead, the temperature inputs were simulated by exact manual adjustment of a threaded piston inserted in the controller housing in place of the thermostat cable.

The oil make-up circuit performs two functions. First, it supplies oil from a gerotor pump to the ball piston pump pintle to "make up" for journal leakage. Secondly, it provides for fan system cooling and filtration.

Check valves have been incorporated in parallel to both fan motors to prevent overpressurization in the return line during fan deceleration.

5.2 Component Description

5.2.1 Pump

The fan drive pump is basically the 8.7 in^3/rev variable displacement radial piston ball pump designed in 1966 for the XM-1 hydromechanical transmission. The housing, pintle, and actuators were designed to conform to fan drive packaging criteria (e.g. PTO flange mounting and integral sump) and to be adaptable to a reversed direction of input shaft

rotation. The major elements of the pump are a rotating cylinder block, a stationary porting pintle, a strokable race, stroke actuating pistons, an integral make-up pump, and a relief valve.

The cylinder block is a seven-cylinder radial ball piston design which rotates on a nodular iron pintle journal.

The race is anchored at the pintle pin and pivots on a spherical bushing pressed into an ear on the race. Opposite the spherical bushing is a tang on which actuator forces are applied to hold or stroke the race. Stroking of the race is governed by the controller's hydraulic signal to the large diameter actuator servo piston. The opposing actuator piston receives a hydraulic signal from the pintle high pressure port. The stroke actuator piston in this system is shimmed to produce a modest initial pump stroke. This provides the fan system idle speed.

A gerotor make-up pump circulates oil through the fan drive cooler and filter and back to the low pressure pump pintle port. Oil required to compensate for ball piston and journal leakage enters the pintle; the remainder is passed to sump through the relief valve at the pintle port. The relief valve limits intake pressure to 50 psig.

The pump assembly is adaptable to reversed input rotation by orientation in assembly of the make-up pump porting plate and the actuator housing.

5.2.2 Controller

The purpose of the controller is to regulate fan speed and limit pump output pressure by communicating a stroke signal to the pump servo actuator piston in response to temperature input signals, individual fan speeds,

and pump output pressure. The temperature input signals are mechanical inputs from remote thermal power elements. Fan speed is communicated by sensing the flow across an orifice in both fan return lines.

The controller is mounted directly to the pump pintle and consists of an aluminum housing with a single valve servo relief circuit and a two-valve speed sensing circuit.

The servo relief circuit is normally closed at the servo relief valve until pump pressure reaches a predetermined value (approximately 10% above system pressure at maximum fan speed). At this pressure, the valve spring force is overcome and the relief valve feeds the pump destroking piston causing pump output to decrease. The purpose of this circuit is to prevent overpressurization in the event of fan stall or line blockage and to prevent a gross overspeed of the fans (servo pressure is also a function of fan speed).

The speed control circuit is responsible for controlling fan speed in proportion to temperature input independently of engine speed. This is accomplished by a summing bar and pilot valve used in conjunction with a speed reference valve, thermostat input signals, and a light return spring as shown in Figure 3.

The pilot valve communicates a hydraulic signal to the pump servo actuator piston in response to the mechanical action of the speed reference valve and temperature inputs on the summing bar.

The speed reference valve moves vertically upward in proportion to flow across the fan return line orifice by balancing, with a spring, the

upstream and downstream orifice pressures at the ends of the valve. A shuttle ball check valve is used between the upstream lines to select the highest flow to be controlled. This flow differential between both fan lines is insignificant during normal operation.

The temperature inputs act vertically downward on the summing bar in proportion to the temperature at the thermostat housings.

For each position of the summing bar due to temperature input, there is a corresponding geometric equilibrium position of the speed sensor valve which allows the pilot valve to achieve a balanced (null) position. An independent change of temperature or speed input moves the linkage bringing the pilot valve out of its null position. The resultant pilot valve signal to the pump servo actuator piston adjusts the pump flow to the new input and restores equilibrium. Thus, the system maintains a constant fan speed for a specific temperature input and is unaffected by pump speed.

5.2.3 Motor - Picture or Sketch.

The hydraulic motors are positive displacement radial ball piston motors whos major components are front and rear housings, a twin port pintle, a rotating cylinder block, 5/8 diameter precision steel balls, an elliptical race, and a splined output shaft. (See figure 4) Two variations of this motor were buil for the fan drive system. One was a 1.0 in^3/rev motor with a seven-cylinder block; the other was a 1.25 in^3/rev motor with a nine-cylinder block.

Oil supplied under pressure from the pump is fed to one of the pintle ports, forcing the balls outward against an elliptical race. Part of this outward force is vectorially resolved into a specific torque causing

the cylinder block and output shaft to turn. The oil charged cylinder then surrenders its oil through the adjacent pintle exhaust port by the inward movement of the ball as it runs in the race.

Orientation of the elliptical race in assembly gives the capability of reversing the direction of motor rotation.

5.2.4 Thermostat Assembly

The thermostat assembly shown in Figure 5 has the function of communicating a mechanical signal to the controller to indicate the temperature of the object cooling system.

The thermostat housing is mounted in a cooling line with the sensor end of a thermal power element protruding slightly into the oil flow path. This power element, when coupled to a specific spring reaction force and rate, will produce a mechanical movement in proportion to temperature over a specific range. The Vernatherm in this system activates at 170°F and provides a signal up to 210°F. Its total mean piston travel is .450".

6.0 RESULTS AND DISCUSSION

6.1 Temperature/Speed Characteristics

With the temperature input to the summing bar accomplished manually, a steady-state performance matrix was constructed by plotting fan speed at various increments of controller input and pump speed. System response during the test was smooth, accurate, and repeatable. The matrix for the 6000 rpm fan system is shown in Figure 6. A graph of fan speed versus temperature input at constant pump speed is shown in Figure 7.

An analysis of Figure 7 shows that the fan control system is essentially a dual stage system. The first stage is the fan idle mode which allows the fans to turn at low speed when cooling is not required. The fan idle speed, being a function of initial pump stroke, changes slightly in proportion to pump speed. With pump speeds of 900 to 3200 rpm, the idle speed varied from 105 to 340 rpm.

The second fan control stage begins when the thermostat input (or manual actuation) begins to move the controller summing bar. At this point the fan speed rises quickly and smoothly to approximately 2100 rpm. (The thermostat input to signal this transition should be designed such that it occurs at 170°F.) Any further incremental input to the summing bar results in a corresponding change in fan speed. An input of .325 inch will increase fan speed from 2100 rpm to 6000 rpm.

The horizontal nature of the isothermal fan speed lines demonstrates the system independence of pump speed.

The maximum overdrive ratio line establishes the boundary beyond which the pump is unable to supply sufficient output to maintain a high fan motor speed. The position of this line in the matrix in no way hinders the cooling capability of the system because the highest cooling requirements generally occur at relatively high engine speeds. Full cooling capability is available at approximately 2100 rpm PTO speed.

A temperature/speed matrix was also plotted for the 8000 rpm fan system and is shown in Figure 8. With this system the idle speed varies from 110 to 360 rpm over the range of pump speed. The threshold fan speed is 3100 rpm and .325 inch of summing bar stroke is required to increase the fan speed to 8000 rpm.

6.2 Fan Speed Balance

For both the 6000 rpm and 8000 rpm dual fan systems, a series of tests was run to determine if one fan speed would differ significantly from the other. The results of that testing are shown in Figure 9. In summary, even though high hydraulic line length ratios were used between the individual fans, the difference in fan speed was judged to be insufficient to warrant the use of flow divider devices.

The relatively equal fan speed performance is due to the fan's torque/speed relationship. Fan speed varies with the square root of input torque. Therefore, for any fan motor torque reduction due to increased line loss, the effect on fan speed will be attenuated by this square relationship. The uniformity of reaction of both fans to dynamic input is demonstrated in Figure 10.

6.3 Fan Overspeed

Analysis of the fan drive system shows that fan overspeed is not a problem and additional speed override devices are not required. Within the controller, the speed of the fastest fan is automatically controlled and the servo relief valve setting places an absolute ceiling on fan speed.

For such a fan system, fan overspeed was thought most likely to occur as a result of excessive temperature input to the controller, high pump acceleration rates at maximum pump stroke, or the stalling of one fan. Extensive testing indicates that none of these areas present a serious problem.

Sensitivity to increased pump stroke is shown by the position of the dashed line on Figures 6 and 8. For both fan systems, maximum fan speed (6000 or 8000 rpm) is attainable with .325 inch temperature input. Speeds of 6600 and 8800 rpm are possible with .050 inch additional travel. (NOTE: The capability of the system to allow 10% overspeed was an initial system objective.) Although higher speeds were not plotted, the system will control to higher fan speed. For this reason, the actuating thermal elements must be carefully matched to the system or a stroke limiter should be incorporated into the thermostat assembly.

Figure 10 shows that under high pump acceleration the fan overspeed is slight (285 rpm) and of brief duration.

Figure 11 shows that the stalling of one fan does have a slight effect on the performance of the other at high speed. An overspeed of 300 rpm was recorded at .325" adjustment.

6.4 Fan Stall

The system is designed so that it cannot be damaged by fan stall. The servo relief valve protects against the stalling of both fans by destroking the pump. The stalling of one fan results in the other fan being controlled by the normal controller speed/temperature schedule.

Testing to confirm the operation of the servo relief valve was clouded by the poor condition of the valve and sleeve after an early incidence of system contamination. Throughout the test the valve would become stuck fast in the sleeve bore.

With one fan blocked, a performance matrix was run with the other fan and is shown in Figure 11. Note that the maximum overdrive ratio has been shifted significantly. This occurs because the pump output to the one remaining fan motor doubles and high fan speeds are attainable at lower pump speeds. The speed/temperature schedule was essentially unchanged from the dual fan system. At the maximum temperature input stroke (.325") the observed maximum fan speed was 6300 rpm. With both fans operating the maximum speed was 6000 rpm.

6.5 Effect of Pump Acceleration

Figure 10 exhibits the reaction of fan speed to rapid acceleration of the pump. This graph shows that both fan speeds behave uniformly and that they are held in a fairly narrow band even though the pump speed is accelerating quickly through a wide range of speeds. With an eye toward possible fan overspeed, it can be seen that with extremely high pump accelerations the maximum fan speed overshoots only 285 rpm (6075 rpm steady-state, 6360 rpm transient).

6.6 Efficiency

The temperature controlled hydrostatic fan drive system allows greater overall system efficiency by allowing the fans to idle at speeds from 100 to 360 rpm when cooling is not required (below 170°F engine and transmission oil temperature). At these fan speeds the pump horsepower requirement varies from approximately 2 HP to 10 HP depending on pump speed. The fan idle speed characteristics are shown in Figure 12.

The power requirements for the full range of fan speeds are shown in Figures 13 and 14. These curves are presented as bands because of the pump input power to pump speed relationship (by the nature of the system, any given fan speed can occur over a wide range of pump speeds and pump input power will increase with pump speed).

The cubic relationship of fan speed and power is illustrated by both figures. For reference purposes, an idealized fan power curve is also shown in both figures.

Values may be compared between the actual curves and the ideal curve to gain a feel for the pump to fan efficiency. It must be remembered, however, that the reference curve is merely a mathematical idealization and is not an actual fan performance curve. Two legitimate points on these curves, however, are the rated speeds of 6000 and 8000 rpm. Based on these points, the pump/fan efficiency is 69% (at 6000 rpm) for the 6000 rpm fan and 53.3% (at 8000 rpm) for the 8000 rpm fan system. The lower efficiency of the 8000 rpm fan is a product of the sensitivity of the wet case motors to high speeds and the large pump/motor size ratio.

6.7 System Performance

Reference to the thermostat assembly output curve of Figure 15 indicates the need for engineering a thermal power element specifically to the requirements of the fan drive system. The thermal power element used provided a continuous output signal from 170°F to 210°F as required; however, its output travel from this range was .450 inch and additional travel to over .500 inch at 235°F was noted. The initial design assumptions for the controller provided for only .375 inch thermostat output with this .375 inch travel allowing a 10% fan overspeed capability (6600 rpm and 8800 rpm). To achieve the rated fan speeds of 6000 and 8000 rpm, an input of .325 inch was required.

Without controller geometry changes, the increased travel of the thermostat signal was inconsistent with the desired fan speed range.

To evaluate the performance of the thermostats in the system, a stroke limiter was incorporated in the thermostat housing to eliminate any travel beyond .450 inch (at 210°F) and sufficient clearance was left at the thermostat/summing bar interface to result in .325 inch travel. This resulted in system activation at 185°F and the performance curve is shown in Figure 16.

7.0 TEST PROCEDURE

All testing was performed in the spin stand facility with power supplied by a 6V53 Detroit Diesel engine. Instrumentation was provided to monitor pump parameters (input rpm, torque, make-up pressure, output pressure, and sump temperature), motor parameters (rpm, inlet pressure, and outlet pressure), and thermostat temperature. The testing procedure follows.

7.1 Green Run

The fans were run at 4000 rpm with pump speeds of 550, 1000, 1250, and 1500 rpm for 30 minutes at each speed. Teardown and component inspection followed immediately.

7.2 Controller Calibration

The fan return line orifices and the speed sensor valve spring force were adjusted to conform to the required speed envelope. After observing the system pressure obtained while running at 6600 rpm (maximum) fan speed, the servo relief valve was set on a flow bench at 10% above that pressure.

7.3 Performance Evaluation

To demonstrate the system temperature/speed characteristics, both a steady-state performance matrix and a dynamic response test were run with the system.

To construct a steady state matrix, fan speed was observed for discrete increments of pump input speed and temperature input signal over their entire ranges. The relationship of summing bar travel and fan speed was observed to aid in the later installation of the thermostat assembly.

The dynamic response test was performed by recording on strip charts the reaction of fan speed to rapid variations of pump input speed.

7.4 Fan Speed Differentials

A complete performance sequence was run to determine if the speed of one fan would vary significantly from the other. Testing was performed with both equal and unequal hydraulic line lengths.

7.5 Stall Evaluation

This test was run by communicating simulated fan stall conditions to the controller and observing the system function. The effect of both fans being stalled was evaluated by confirming the operation of the servo relief valve with the flow to both fans blocked. The effect of a single fan being stalled was evaluated by blocking flow to one fan and running a complete performance evaluation of the other.

7.6 Efficiency

Efficiency data for both the 6000 and 8000 rpm fan systems were taken for each point in their performance matrices. Since the instrumentation system did not allow direct reading of fan torque, the system efficiency was expressed indirectly as the horsepower requirements to drive the fans through their entire speed range.

7.7 System Evaluation with Thermostats Installed

Prior to this section of the test, the temperature input to the controller was simulated by manual actuation. This final system evaluation would involve system control by the actual thermostat assemblies. To accomplish this, the output characteristics of each thermostat assembly

were first determined by plotting the mechanical output as a function of temperature. Knowing this and the required summing bar movement to obtain the full range of fan speed, the thermostat assemblies were incorporated into the system and the system performance parameters (thermostat temperature, fan speeds, and engine speed) were plotted on a strip chart recorder.

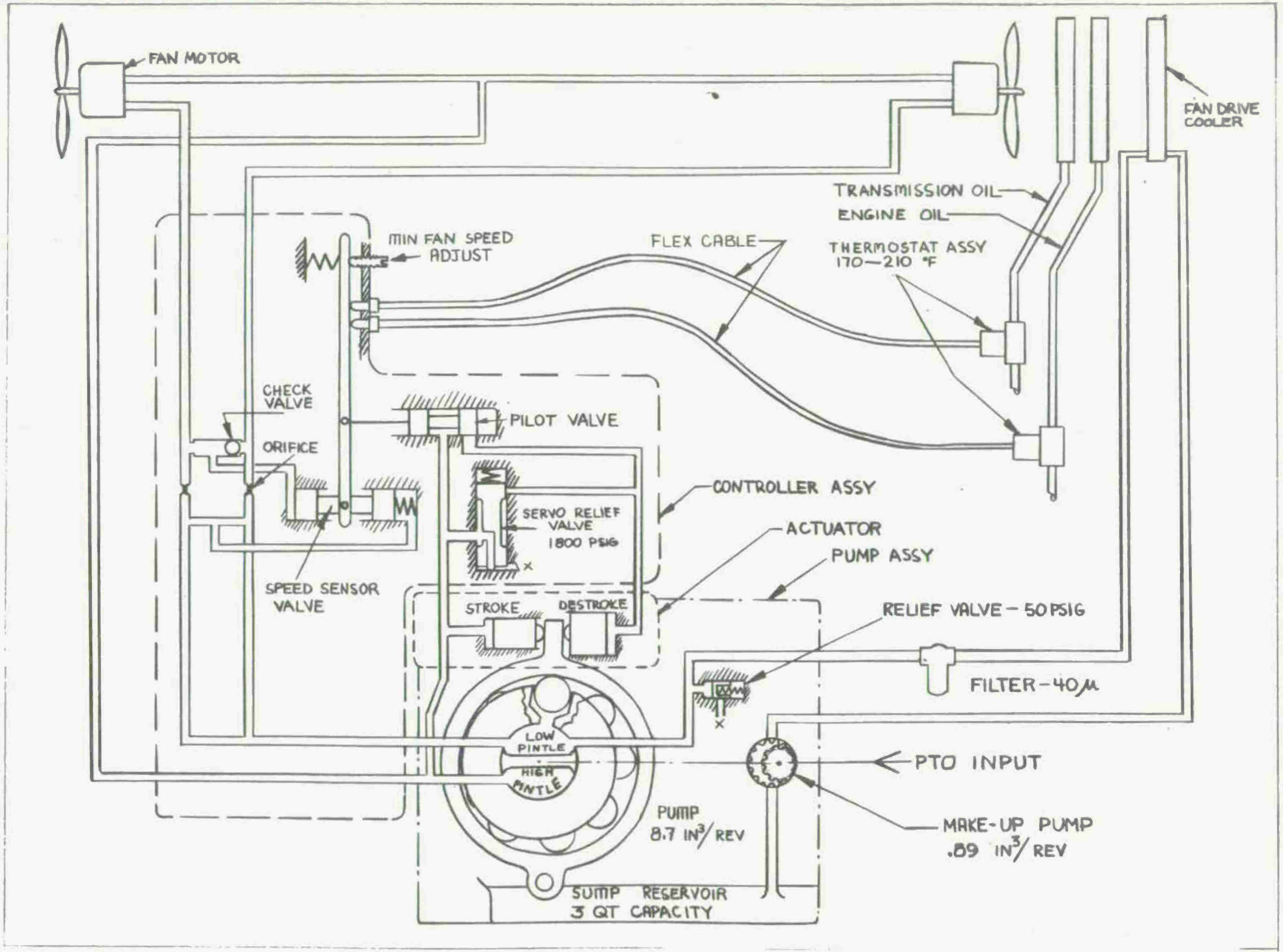
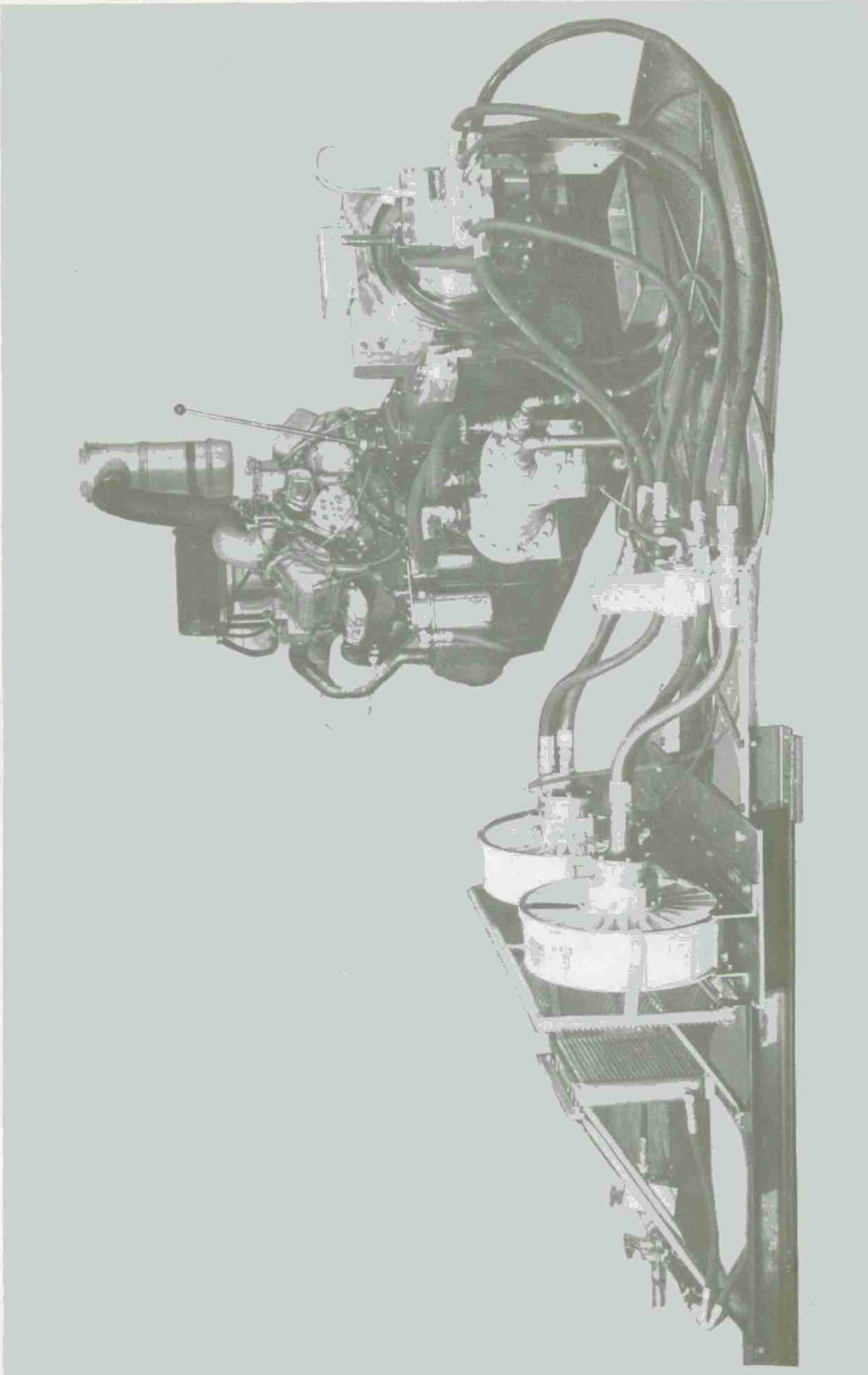


FIGURE 1. FAN DRIVE SCHEMATIC



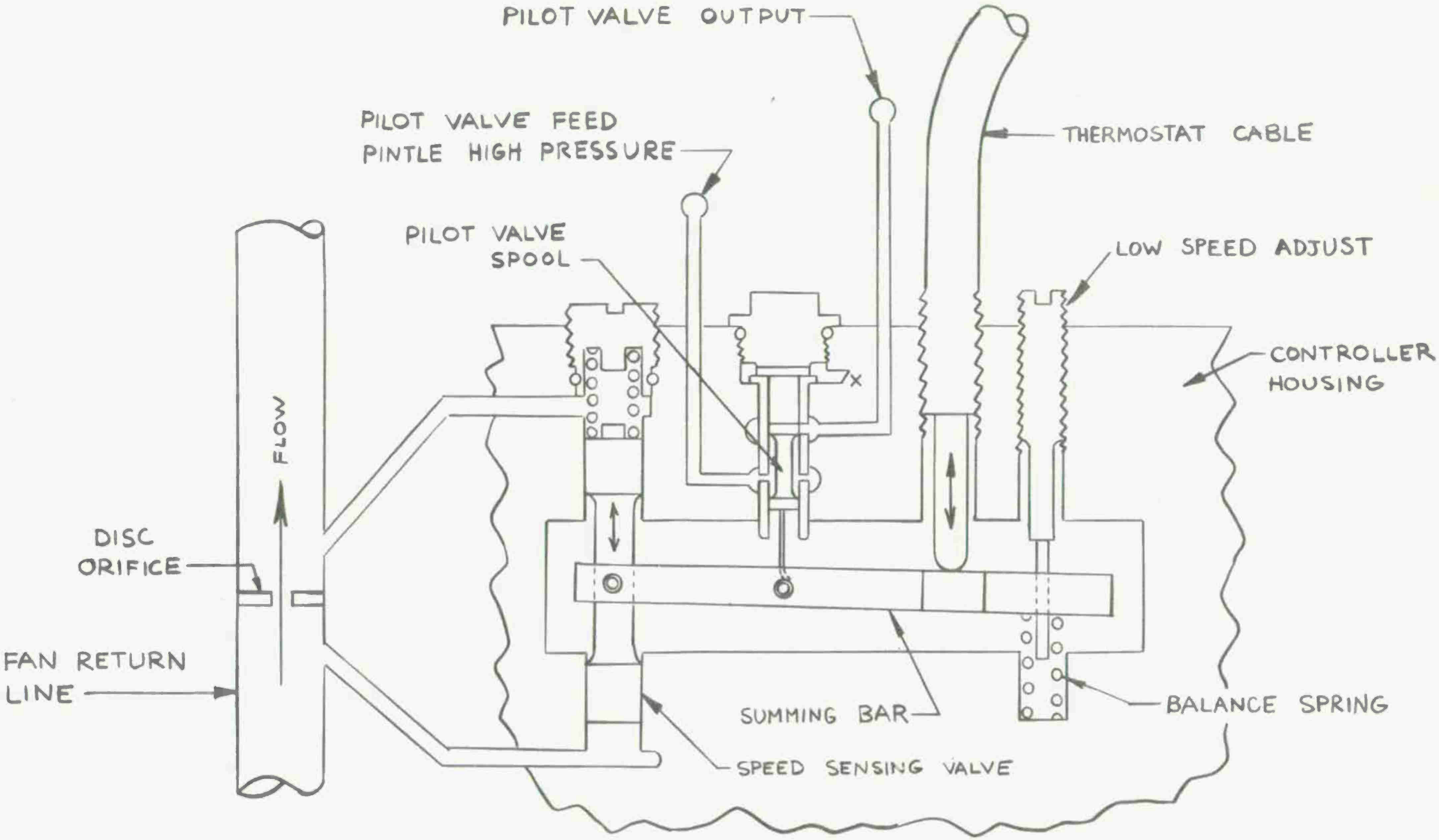


FIGURE 3. FAN SPEED SENSING AND CONTROL CIRCUIT

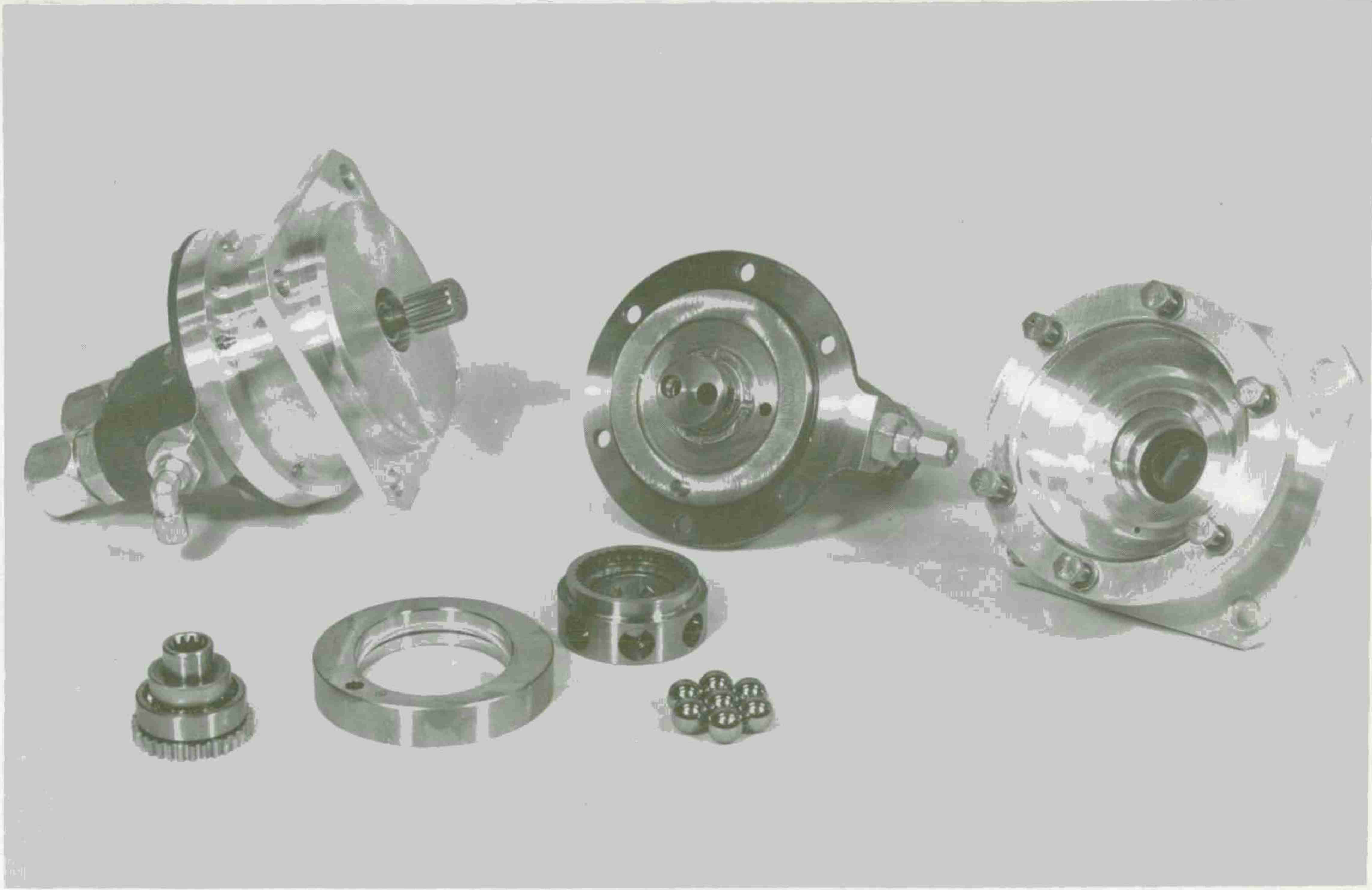


FIGURE 4. MOTOR ASSEMBLY AND COMPONENTS

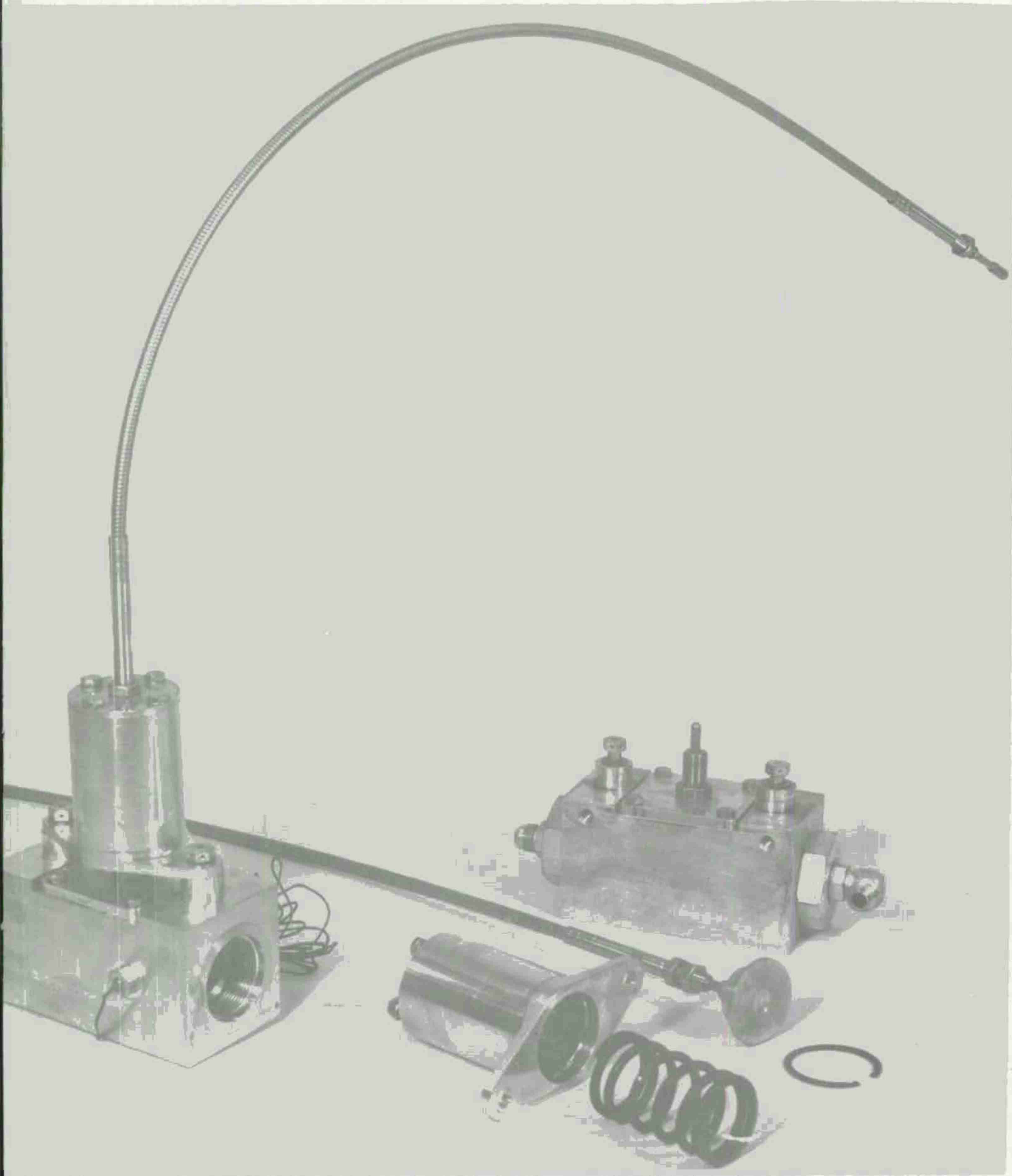


FIGURE 5. THERMOSTAT ASSEMBLY AND COMPONENTS

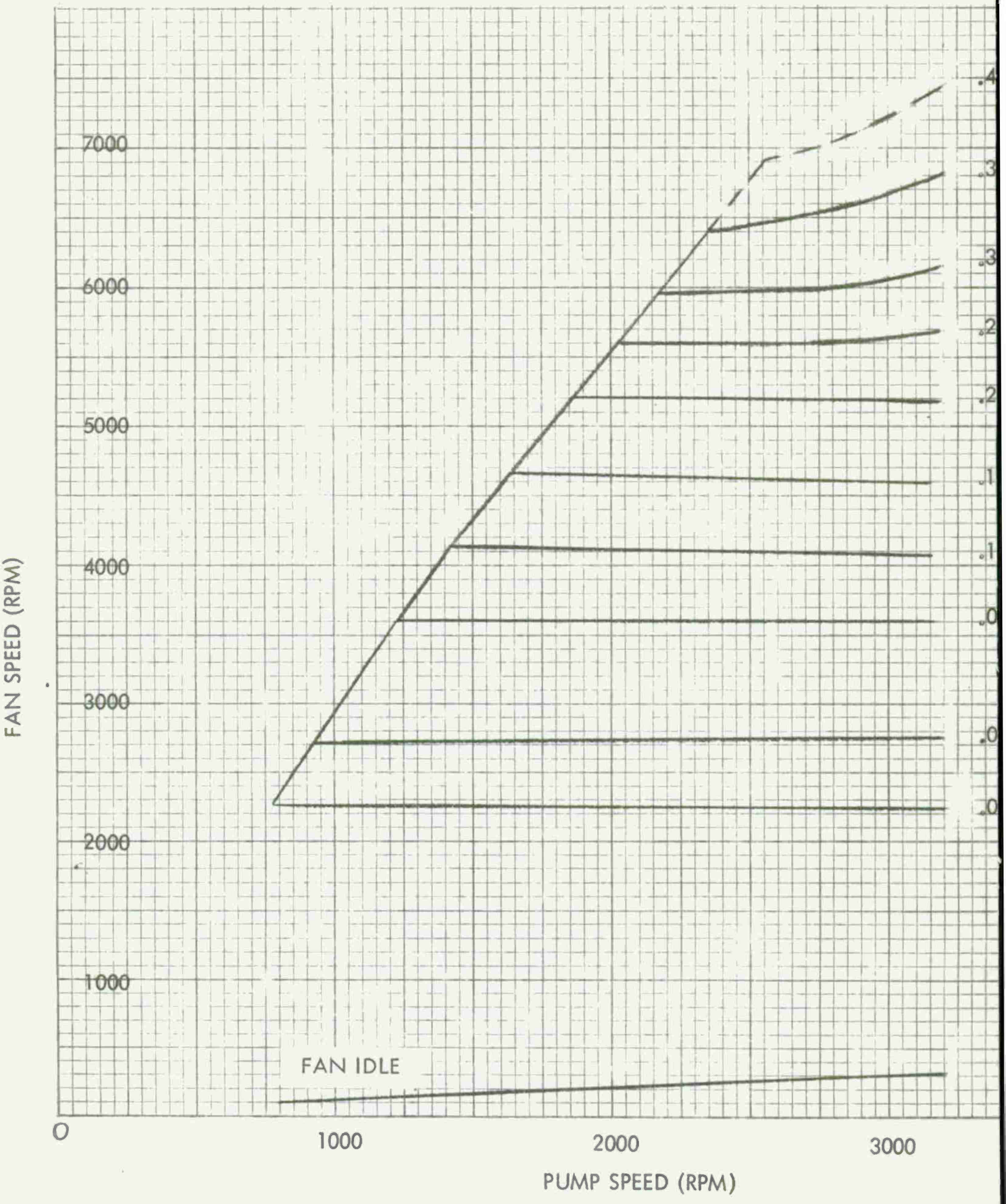


FIGURE 6. PERFORMANCE MATRIX (6000 RPM FAN SYSTEM)

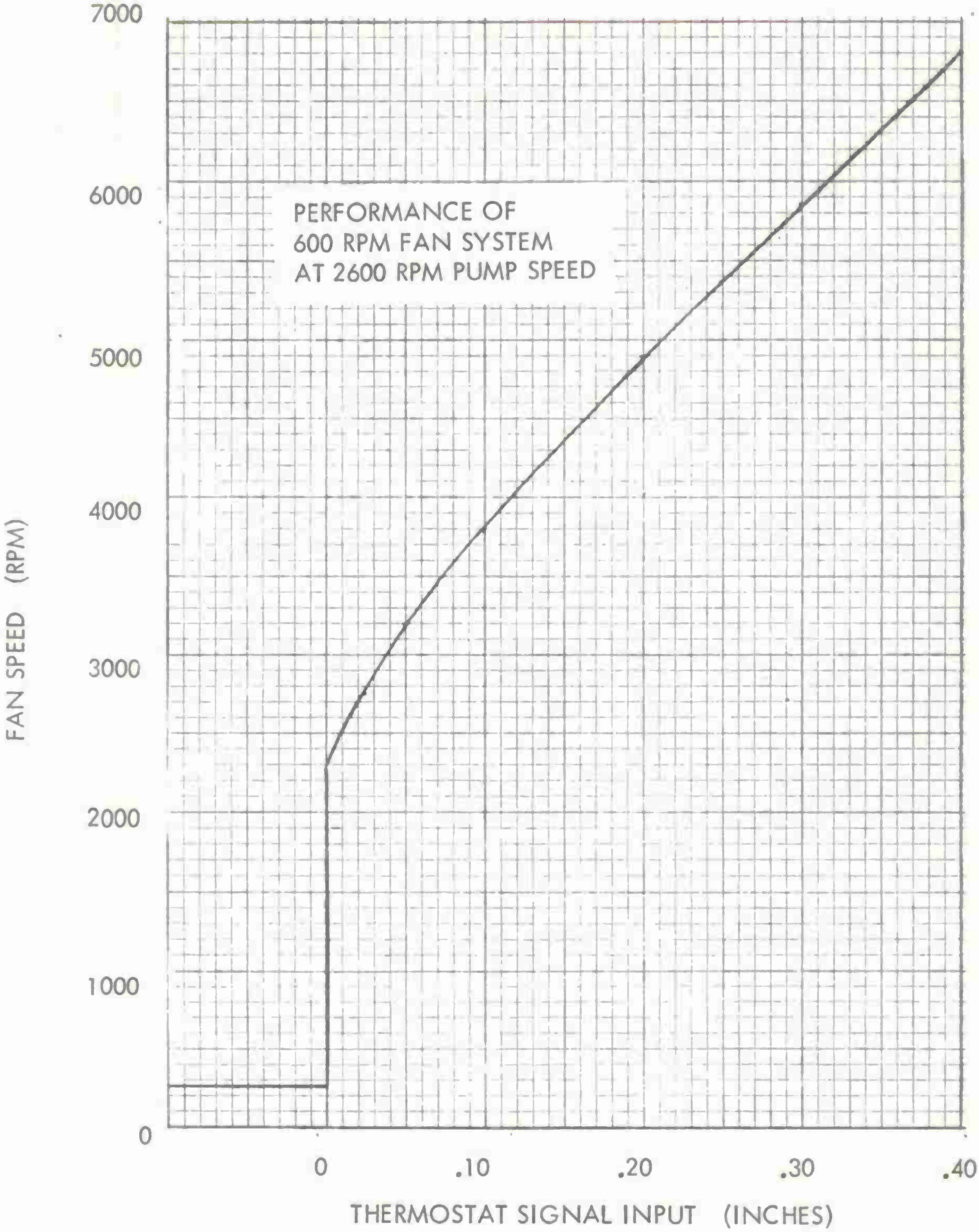


FIGURE 7. SYSTEM PERFORMANCE AT CONSTANT PUMP SPEED

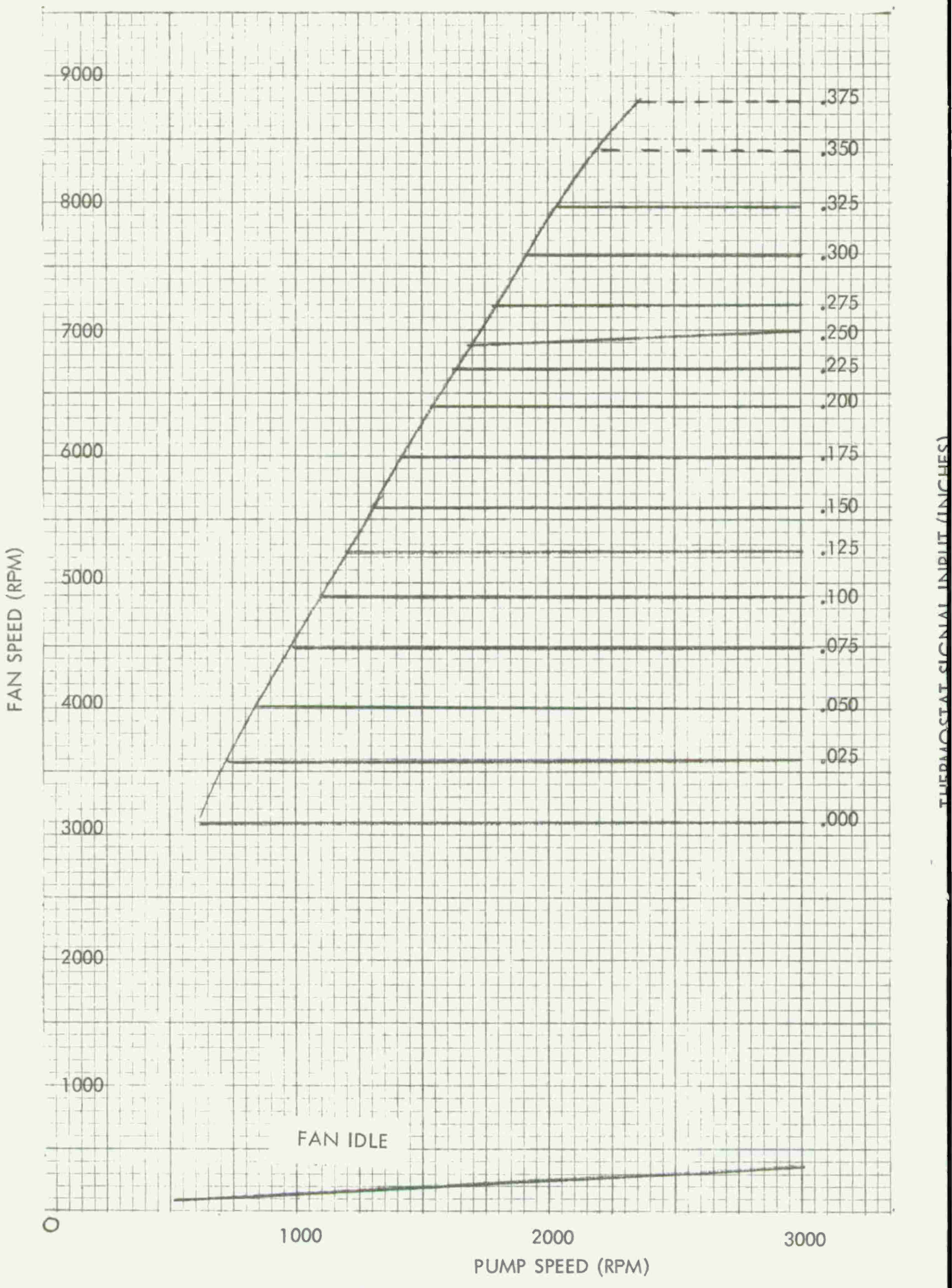


FIGURE 8. PERFORMANCE MATRIX (8000 RPM FAN SYSTEM)

Fan Speed (rpm)	6000 rpm Fan System				8000 rpm Fan System	
	Fan Lines 129" and 69" Length Ratio 1.86:1		Fan Lines 129" and 25" Length Ratio 5.2:1		Fan Lines 102" and 66" Length Ratio 1.55:1	
	Speed Difference (rpm)	% Difference	Speed Difference (rpm)	% Difference	Speed Difference (rpm)	% Difference
2000	10	.5	25	1.25	-	-
3000	30	1.0	45	1.5	30	1.0
4000	40	1.0	30	.75	30	.75
5000	50	1.0	75	1.50	75	1.50
6000	80	1.3	160	2.66	50	.83
7000	-	-	-	-	50	.72
8000	-	-	-	-	100	1.25

FIGURE 9 Speed Variation between Two Fans in Dual Fan System

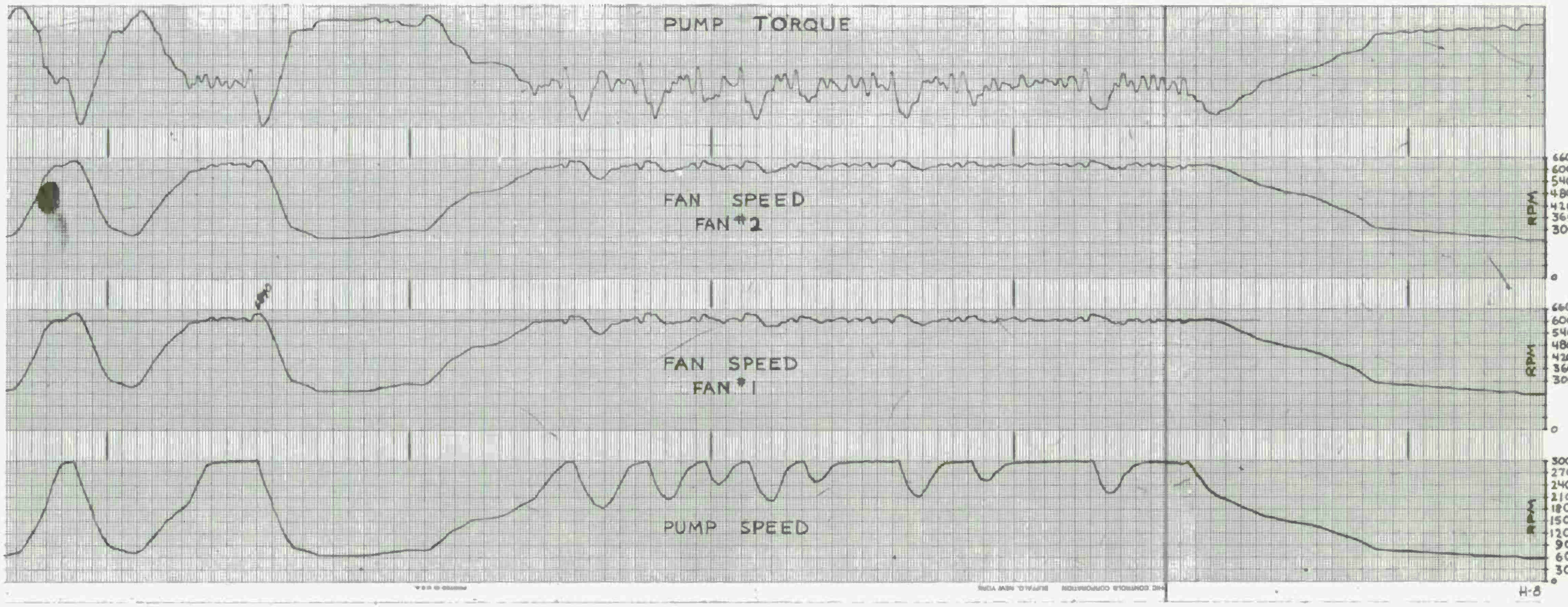


FIGURE 10. REACTION OF FAN SPEED TO PUMP ACCELERATION

6000 RPM FAN SYSTEM MATRIX WITH ONE FAN STALLED

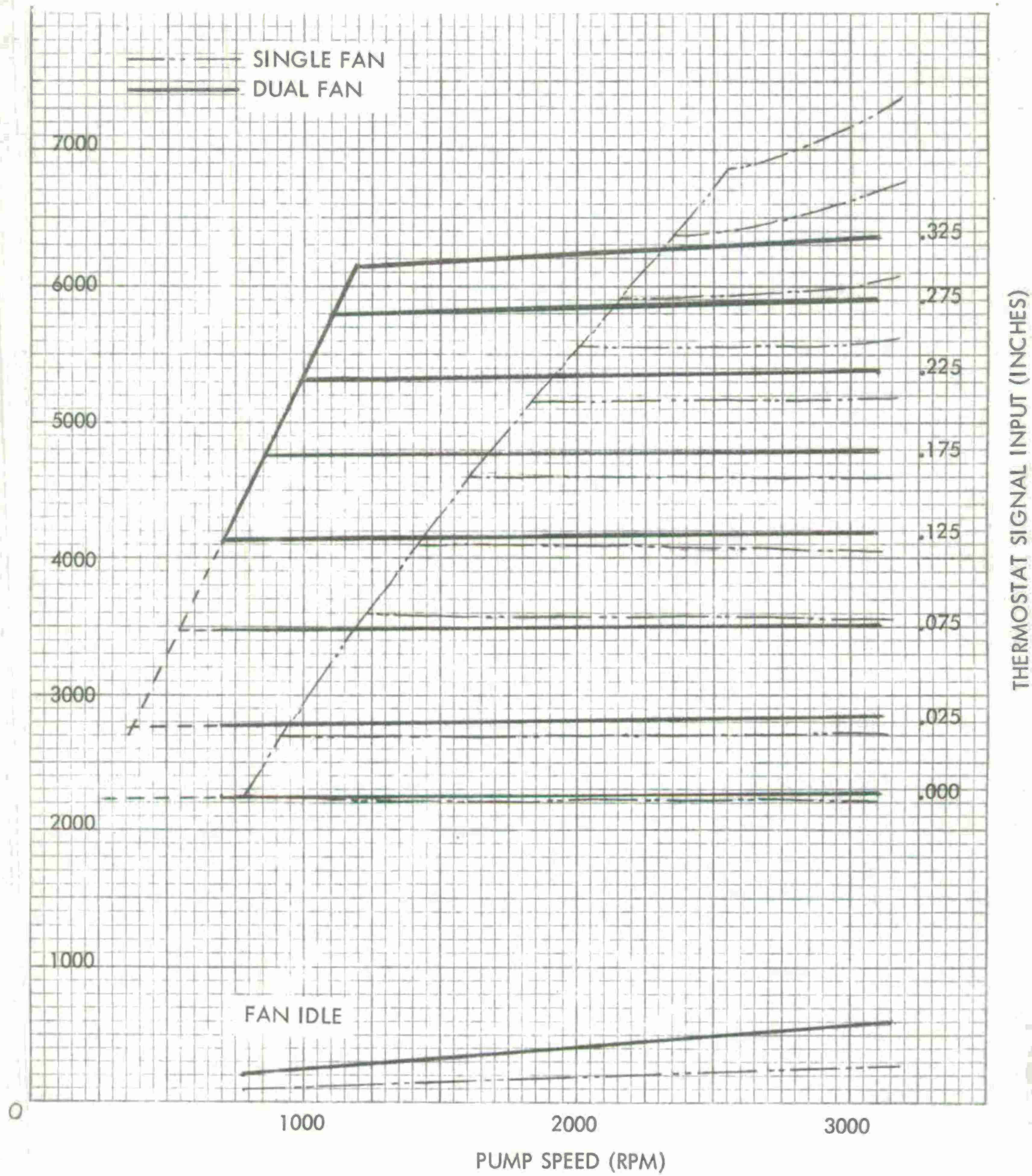


FIGURE 11. SINGLE FAN SYSTEM PERFORMANCE WITH ONE FAN STALLED

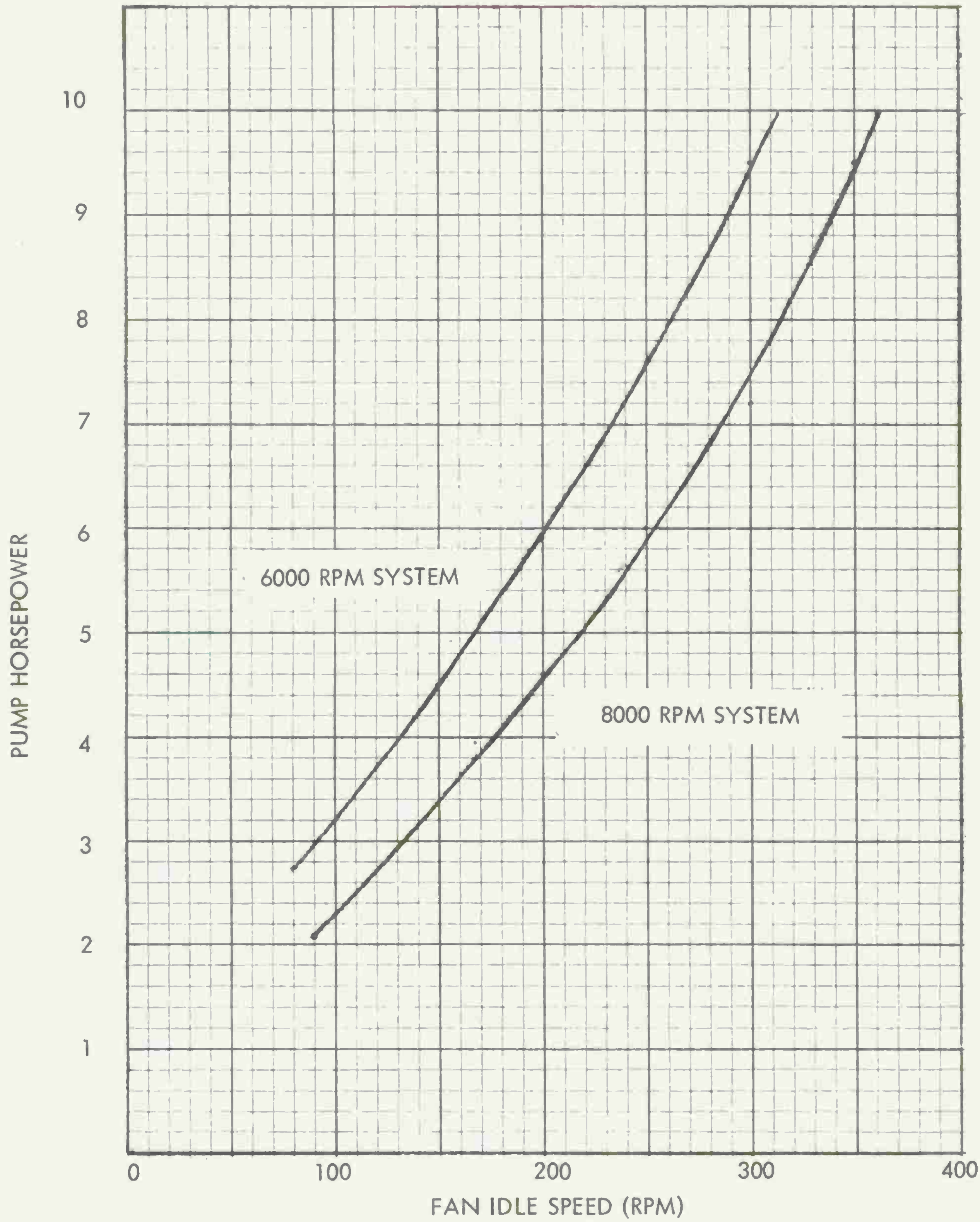


FIGURE 12. PUMP HORSEPOWER REQUIREMENTS AT FAN IDLE

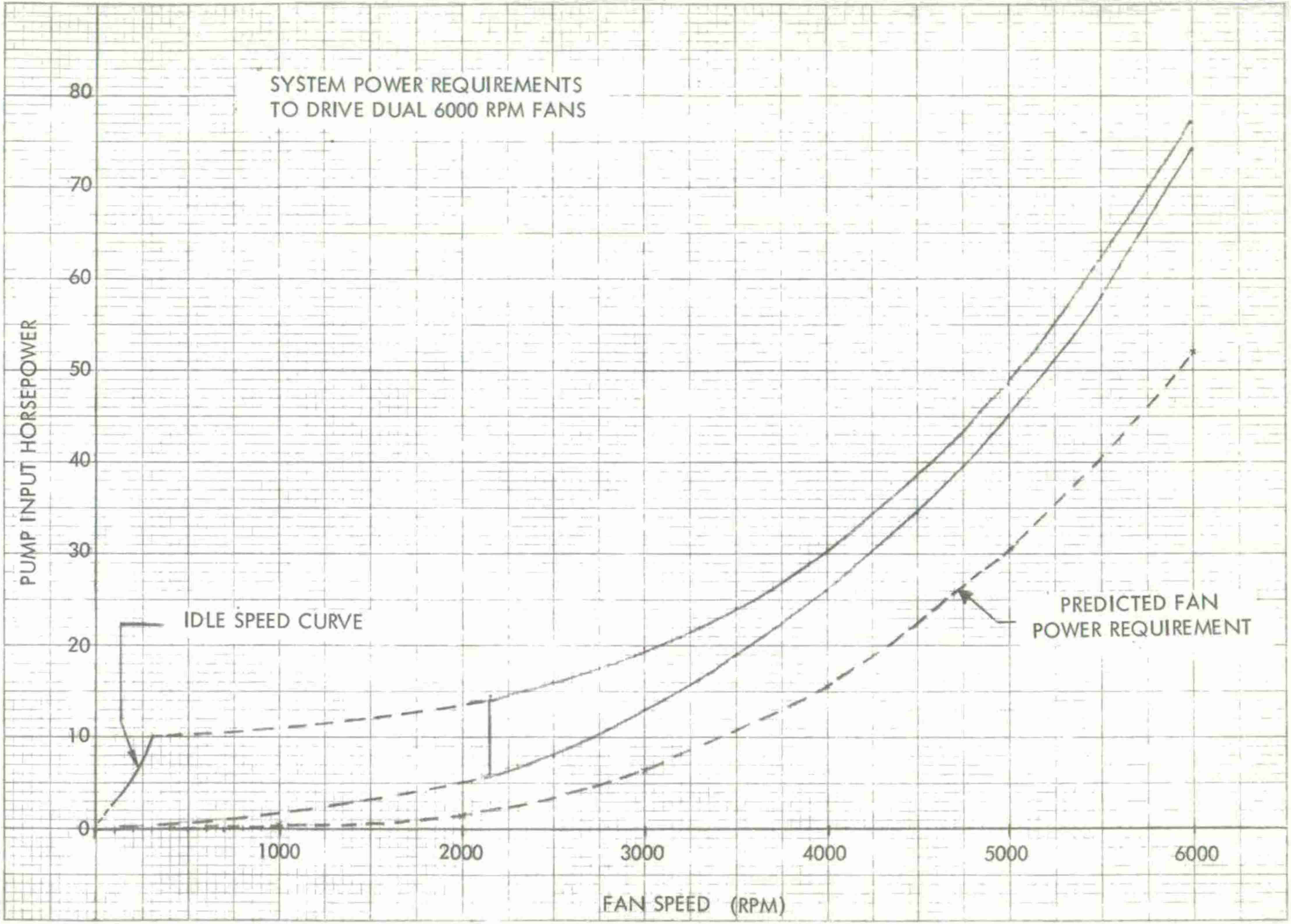


FIGURE 13. SYSTEM POWER REQUIREMENTS (6000 RPM FAN SYSTEM)

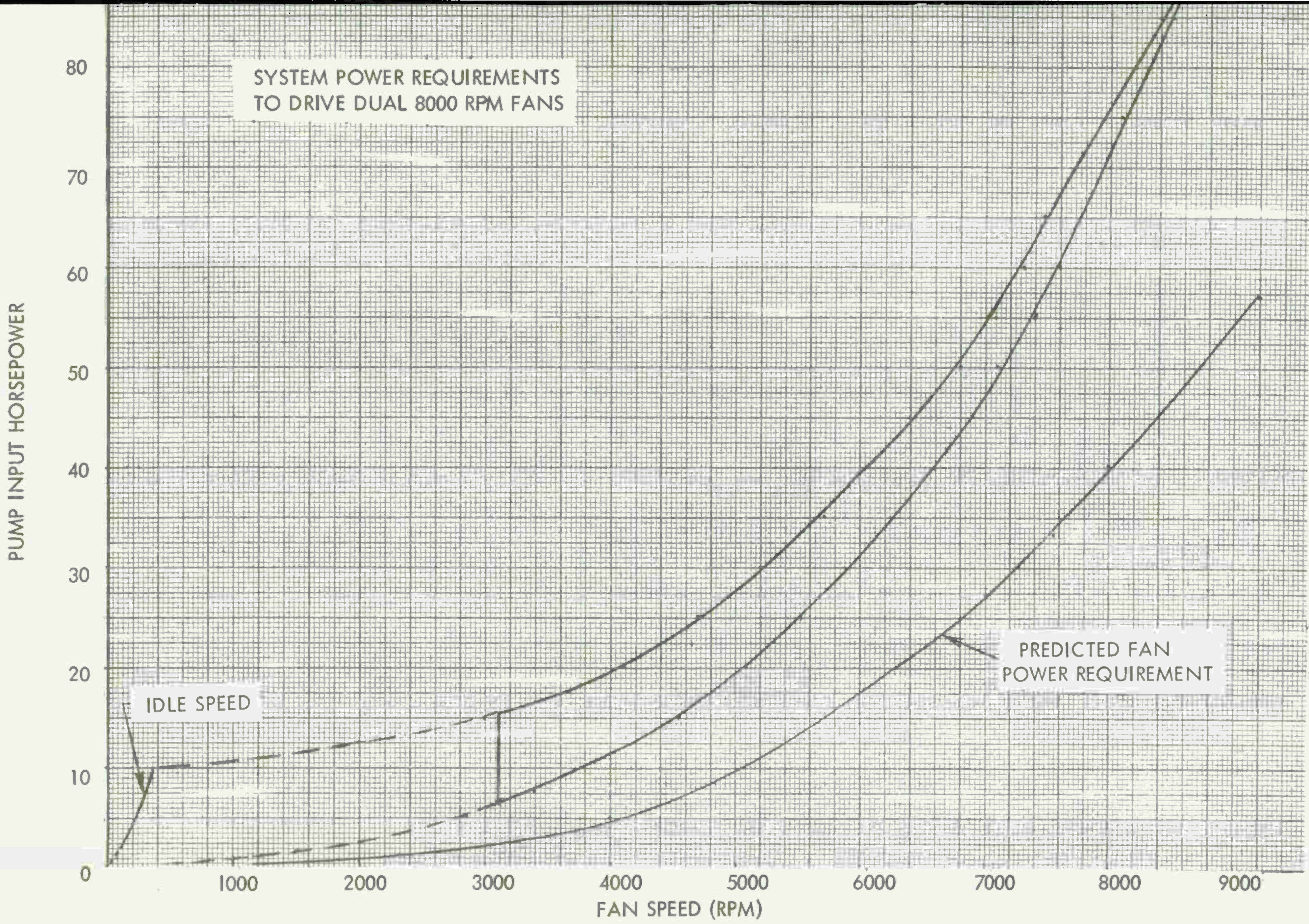


FIGURE 14. SYSTEM POWER REQUIREMENTS (8000 RPM FAN SYSTEM)

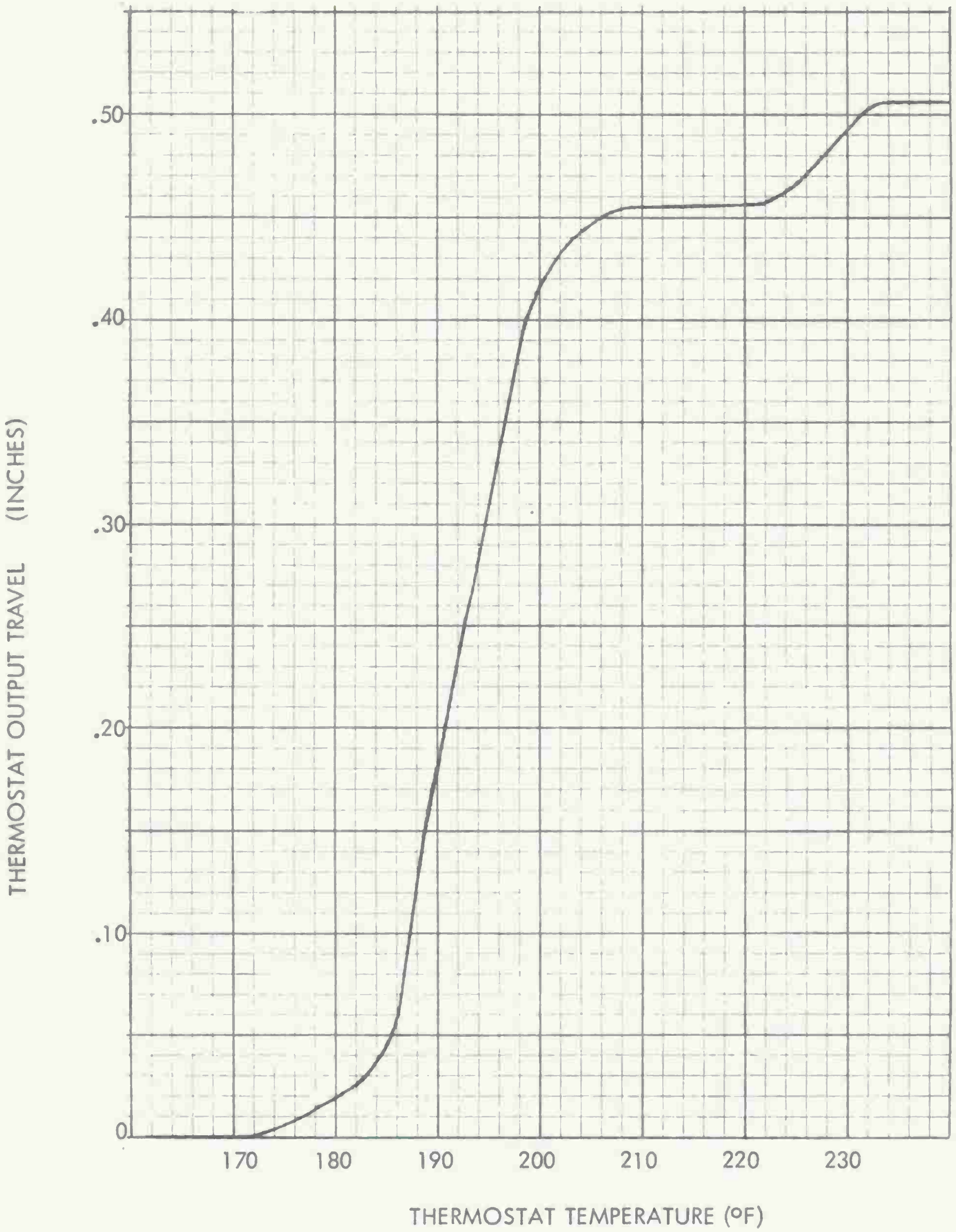


FIGURE 15. THERMOSTAT PERFORMANCE

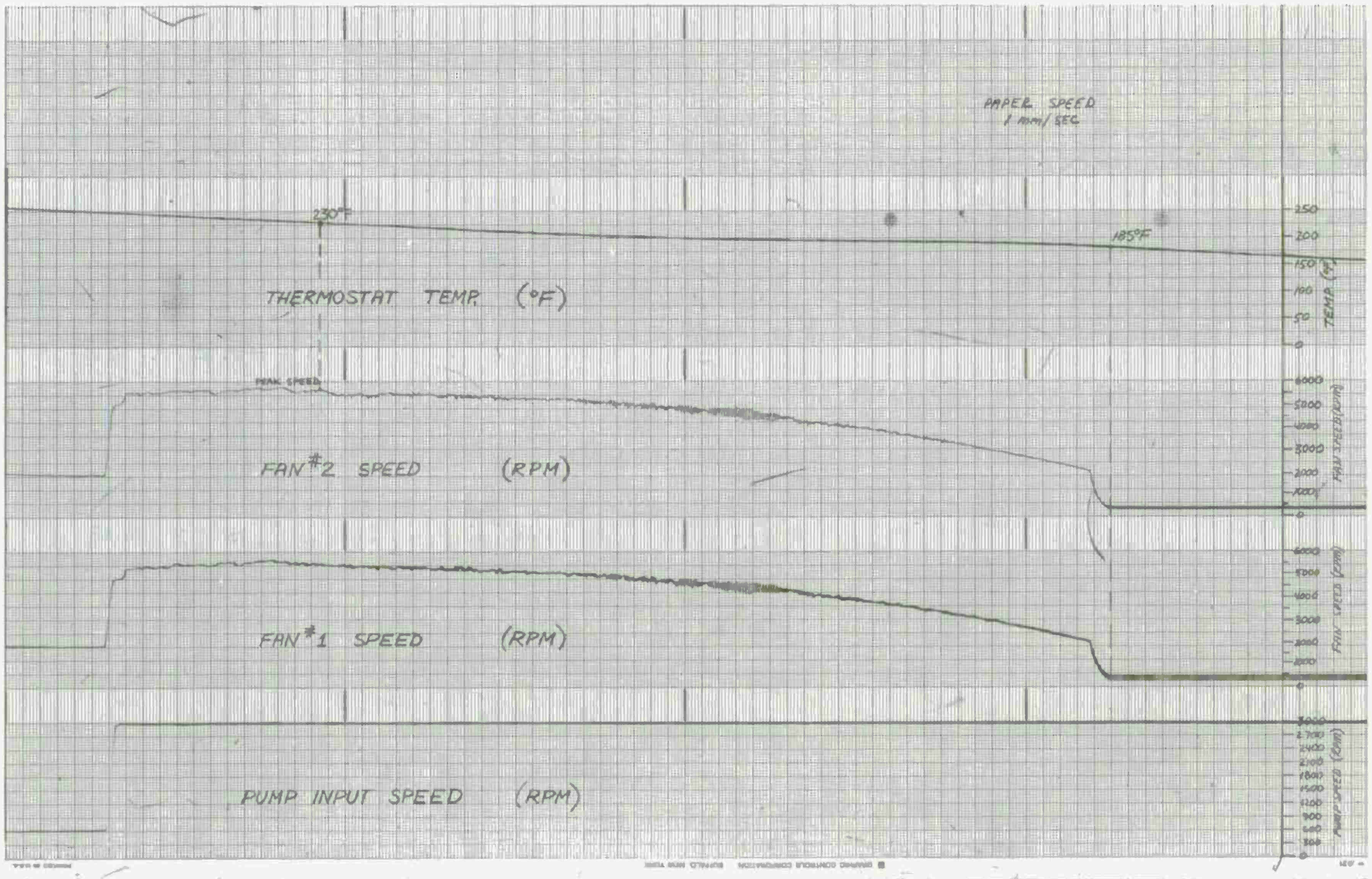


FIGURE 16. SYSTEM PERFORMANCE EVALUATION

APPENDIX

FAN DRIVE SPECIFICATIONS

COMPONENT SPECIFICATIONS

Pump Assembly

- Displacement: 0 - 8.7 in^3/rev (seven 1.625" diameter balls)
- Pressure: 3000 psig maximum
- Speed: 4000 rpm maximum
- Direction of rotation: clockwise or counterclockwise
- Make-up pump: .89 in^3/rev gerotor (14 gpm at 3600 rpm)
- Make-up relief pressure: 50 psig
- Servo relief pressure (ref): 1800 psig (adjustable)
- Mounting flange: eight 3/8 bolts on 7.00 in. diameter bolt circle
- Input spline: 16-tooth 30° P.A.; 16/32 pitch 1.00 P.D.
- Sump capacity: three quarts
- Oil: SAE 30
- Weight: 82.5 lb.

Controller

Hydromechanical, two valves with summing bar and servo relief valve.

- Temperature input: mechanical, variable
- Hydraulic input: pump pintle pressure; fan return line flow
- Output: hydraulic signal to pump actuator servo piston
- Servo relief pressure: 1800 psig (adjustable to suit fan speed requirements)
- Weight: 9.8 lb.

Motor

- Type: fixed displacement radial ball piston double lobe

- Displacement: 1.25 in^3/rev, 9 cylinder, 5/7" ball
- Rated speed: 6000 rpm
- Rotation: clockwise or counterclockwise
- Mounting flange: four ½" bolts on 6.375" bolt circle
- Output spline: 13-tooth, 30°P.A.; 16/32 pitch .8125 P.D.
- Weight: 13.7 lb.

Motor (8000 rpm)

Same as above except:

- Displacement: 1.00 in^3/rev, 7 cylinder, 5/8" ball
- Rated speed: 8000 rpm

Thermostat Assembly

Aluminum housing with thermal power element.

- Power element: .450" travel from 170°F to 210°F
- Reaction spring: 120 lb. initial load; 115 lb/in rate
- Control cable: push-pull precision ball bearing cable, 1.0" stroke, 4" bend radius
- Assembly weight: 7.6 lb.

Cooler

- Type: oil/air
- Heat rejection: 2257 BTU/min
- Oil outlet temperature: 208°F
- Air static drop: 5.3" H_2O
- Oil pressure drop: 6.3 psi
- Flow H.P. air: 3.23

Filter

- 10 micron nominal
- 15 micron absolute
- 20 psi pressure drop at 14 gpm

Check Valve

- Opening pressure: 5 psig
- Proof pressure: 5000 psig

Hose and Fittings

- High pressure hose: 1.0 inch I.D.
- Low pressure hose: 1.0 inch I.D.

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